SUBSYNCHRONOUS VIBRATION PROBLEMS IN HIGH-SPEED, MULTISTAGE CENTRIFUGAL PUMPS

by

Ian C. Massey

Director of Engineering

Pacific Pumps Division of Dresser Industries, Incorporated

Huntington Park, California

Ian Massey is Director of Engineering of the Pacific Pumps Division of Dresser Industries, Incorporated. Prior to joining Dresser he held engineering positions in the pump businesses of Wormald International and Mather & Platt, Limited.

Educated in the United Kingdom, he received the degree of Master of Engineering from the University of Sheffield. He is a member of the Institution of Mechanical Engineers, and registered as a chartered engineer in the United Kingdom.

Mr. Massey has authored several papers associated with pump design, and the financial justification of computer aided engineering. He has also contributed to books on the subjects of pumps and CAD/CAM.

ABSTRACT

Subsynchronous vibration problems were experienced on two reactor charge pumps during on-site commissioning, which led to large whirl amplitudes, making the units automatically trip. A number of possible mechanisms of excitation were investigated, and the problem was identified to be associated with the influence of the fine annular clearances in the machines on the stability of their rotors. The problem was rectified by making modifications to the annular seal configuration. Following an experimental test program, the machines were successfully recommissioned.

A comprehensive analytical investigation into the stability of the rotor for varying seal configurations was carried out in parallel with the experimental test program. Agreement between analysis and test data was found to be good.

INTRODUCTION

Two high speed, multistage pumps designed for reactor charge duties failed within one hundred hours of intermittent operation following startup of an extension to a refinery in California. Site data captured by refinery inspectors during the plant startup identified that the failures were associated with a subsynchronous vibration at a frequency equal to 70 percent of the running speed. The internal rubbing, which resulted from the whirl orbits of the rotor, generated wear in the pump internal seals, doubling the running clearances in the short period of operation.

The identification of a further subsynchronous vibration on two pumps of similar design, but with a reduced number of stages and allocated to a different duty, resulted in a delay of the startup of the refinery extension, pending the development of a solution to the problem, and subsequent modification of the machines to make them suitable for service. To achieve this, a comprehensive analysis of all possible causes of the rotor whirl was embarked upon by both the pump vendor and user—this resulted in a test program to validate the final analysis and modifications.

The coordinated effort of the two parties resulted in the expeditious modification to the pumps, which have now clocked 6000 hours of running time without any sign of the original problems.

THE PUMP CONFIGURATION AND DUTY

The pump design comprises an eleven stage barrel case design (Figure 1) driven by an electric motor through a gearbox at 6600 cpm pumping 470 gpm of hydrocarbon at a maximum temperature of 523°F. The hydrocarbon average specific gravity is 0.775 with a viscosity of 0.70 cp.

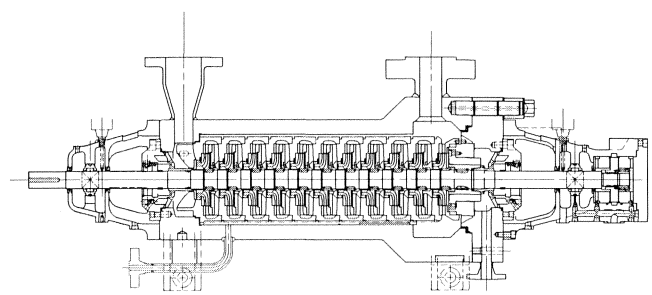

Figure 1. Eleven Stage Barrel Case Pump.

During the plant startup, the pumps had only operated in the recirculation mode, and the maximum temperature achieved was approximately 400°F prior to failure. The pump design had been used previously in the same stage configuration operating at 87 percent of the refinery duty speed, pumping water for injection duties. The injection pumps (20 in number) had exhibited reliable performance over many years. The shaft diameter under the impellers, however, had been increased from 2.75 in to 3.1 in, to accommodate the higher speed duty. Additionally, the internal clearances in the machines supplied for the injection duties were lower. The clearances in the refinery pumps were dictated by the American Petroleum Institute (API) 610 specification.

SITE VIBRATION DATA AND FAILURE MODE

On initial startup, a low level of subsynchronous vibration was observed at the non-drive end shaft displacement transducers. As time progressed, and the temperature of the pumpage increased, the subsynchronous vibration component increased, as shown in Figure 2, to a point where it eventually swamped the synchronous signal and subsequently replaced the synchronous signal as the dominant mode of vibration.

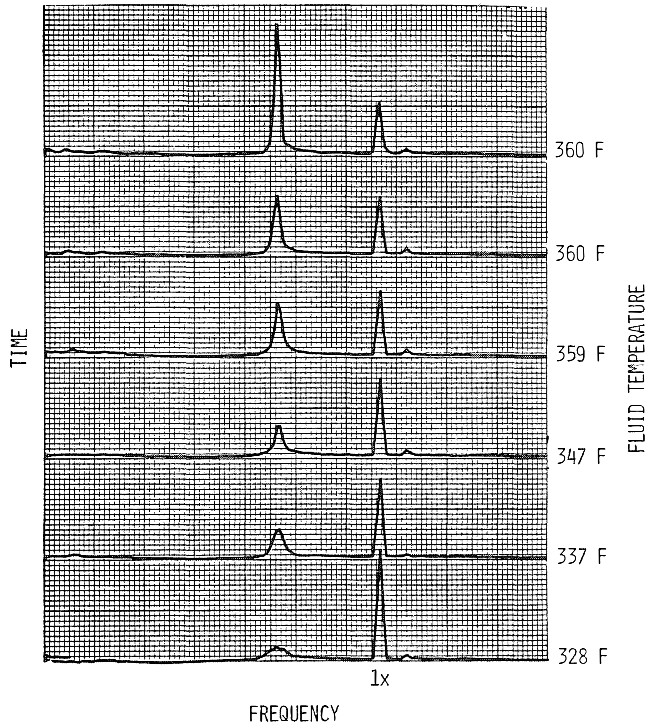

Figure 2. Subsynchronous Vibration Development—Site Commissioning.

Successive starts exhibited similar machine behavior until internal wear allowed the shaft orbit to develop vibration amplitudes which exceeded the trip setting of the vibration monitors, thus tripping the unit before full speed was attained. When the power was removed from the motor, the pump subsynchronous mode showed a frequency variation which was speed dependent, before dropping out at approximately 90 percent speed. Initially, this fact was disguised by the period of the vibration analyzer, which appeared to indicate that no speed-frequency relationship was present. The speed-frequency dependency was confirmed later by the refinery inspectors, following a comprehensive review of the available site data.

Inspection of the unit following the trip on high vibration identified severe internal wear affecting the annular seals in each stage of the machine, and similar wear on the balance drum. The first machine showed normal marking in the tilt-pad journal bearings; however, the second machine, which exhibited less wear internally (due to the shorter running period), exhibited evidence of whirl in the journal bearings. The wear was reasonably evenly distributed circumferentially in the annular seals. During disassembly, no other problems were identified.

The internal wear was heavy, with the pump clearances worn from a nominal 0.022 in on diameter to 0.040+ in on diameter. However, in spite of the heavy wear, no evidence of metal galling was observed. Fortunately, all major components were salvageable, thus reducing the turnaround time once the rebuild configuration had been determined.

ANALYSIS OF THE PROBLEM

A review of the literature, and past experience, identified a number of possible mechanisms which may lead to subsynchronous whirl in a centrifugal pump. However, instances associated with pumps are scarce in the literature. It was necessary to draw on cases identified in other rotating machines. The most common mechanisms are listed in Table 1 [1,2,3,4,5], with comments assessing their applicability to the problem. The most likely causes, which were investigated further, were:

- Rotor instability
- Impeller rotating stall
- Rubbing
- Acoustic resonance in the machine
- Structural resonance in the machine or associated pipework

Table 1. Subsynchronous Vibration Mechanisms.

Mechanism	Frequency Range	Comments
Rotor instability	Subsynchronous Synchronous	Cause of problem investigated
Impeller rotating	Subsynchronous 0.7-0.8 N [1]	Possible mechanism
Diffuser rotating stall	Subsynchronous <0.5 N [1]	Frequency out of range experienced on site
Rubbing	Subsynchronous [2]	Possible cause but eliminated during test program
Acoustic resonance	Subsynchronous Synchronous Super synchronous	Eliminated by calculation and response testing
Structural resonance	Subsynchronous Synchronous Super synchronous	Eliminated by calculation and response testing
Bearing whirl	Subsynchronous 0.15-0.85 N [3]	Journal bearing affects second order in case of problem rotor
Trapped fluid	Subsynchronous 0.5-0.85 N [4]	No significant pockets except gear coupling. Eliminated by inspection
Internal Hysteresis	Subsynchronous [5]	Construction of rotor modified to eliminate possibility

A comprehensive review of the test tapes, which were recorded on the test-strand prior to shipment of the pumps, identified minor traces of subsynchronous vibrations at a very low level which were not identified at the analyzer sensitivity initially used. This effectively eliminated the site as a probable cause, although the experience reported by Corley [6] left some possibility of a predominating site influence. The elimination of acoustics as a primary mechanism was determined by the application of simple equations to calculate the organ pipe frequencies in the passages of the machine:

$$a = \frac{1}{\sqrt{\frac{\gamma}{g}\left(\frac{1}{K} + \frac{d}{tE}\right)}} \qquad f = \frac{a}{L}$$

All modes were calculated to be above the normal excitation frequencies.

Similarly, no structural modes could be calculated below the running speed, and actual measurement of the natural frequency of the bearing housings was made at an early stage of the investigation.

With the simple elimination of acoustic and structural resonances as a primary mechanism, efforts concentrated on the analysis of the stability of the rotor system in the machine. Although a resonant response analysis had been carried out at the design stage, this had been some three years earlier. Some limitations associated with the original analyses were identified to be:

• Both rotating and stationary wear surfaces were grooved with a threaded configuration, making development of accurate stiffness and damping factors difficult. In the case of the balance drum, the rotating surface was threaded and the stationary surface plain. The theory by Black and Cochrane [7], used in the original analysis, is now considered to be suspect for such configurations. Such a view is supported by Wachel [8].

• A true stability analysis was not carried out and the impeller diffuser interaction forces of the type described by Hergt and Krieger [9] and Brennen, et al. [10], were not included in the analysis, due to the difficulty in dealing with the added mass terms. In some cases, these may be modelled as negative stiffness values related to the position of the impellers on the shaft system. It should be noted that access was not available to a program which allowed the added mass term associated with impeller/diffuser interaction forces to be added into the analysis at the time this problem was initially being investigated.

Due to the limitations cited above, and the need to develop a speedy analysis, discussions were held with the California Institute of Technology [10] and Texas A&M University [11] to obtain the most accurate input available, to develop numbers associated with impeller/diffuser interaction forces, and to determine the stiffness and damping factors associated with the fine internal clearances in the machines. Accurate measurements were available from the site, and through the seal stiffness and damping values developed, the site response was simulated, using the computer.

The seriousness of the problem did not allow the manufacturer's engineers the luxury of simulating the site dynamics first, and then theoretically searching for a solution before implementation. While the computer simulation was in progress, modifications were applied to the machines which would maximize the stability of the rotor system. The possibilities considered were:

• increase the stiffness of the shaft system by increasing the shaft diameter, reducing the bearing span and reducing the mass of the shaft mounted components.

• improve component balance and assembly techniques to reduce excitation through mass eccentricity effects.

• increase the impeller/diffuser clearances to reduce interaction forces.

• modify all internal annular seals in order to improve the values of stiffness and damping in the pump fine clearances.

It was only possible, within the constraints of the machine design and hydraulic requirements, to make a minor modification to the shaft stiffness by increasing its diameter. The thick impeller shrouds were machined to marginally reduce the weight of the rotor mounted components. In addition, the impeller-to-shaft and balance-drum-to-shaft fits were carefully analyzed to ensure that the components were still in contact at full speed, yet involved the minimum initial interference fit to effectively control repeatability of the balance of the rotor. This involved the accuracte computation of the impeller displacement under operating conditions, using finite element analysis techniques. The applied balancing procedure involved balancing each component on a mandrel using two-plane balancing, followed by assembly of the rotor, and a check balance of the complete rotor. No adjustments were applied to the assembled rotor. To satisfactorily achieve the required balance, some components required minor redesign to achieve satisfactory repeatability.

The impeller/diffuser clearance could not be modified through the requirement to maintain the design head from the pump. As a result, attention was applied to the internal clearance configuration in the machine.

The pump units were designed to comply with API 610, Sixth Edition. As a result, clearances between the impeller wear rings and the casing rings were 0.022 in minimum on diameter for the application temperature. The high clearances were considered a limit to the flexibility that the designer had to modify the clearance configuration to achieve satisfactory levels of stiffness. Experience over the years indicated that finer clearances yielded satisfactory life at temperatures well in excess of those experienced in this application. Gopalakrishnan [12] provided data indicating the effect of clearance on the shaft stiffness, and the general relationships between clearance, stiffness and damping values were well known in the pump industry. In this case, little time was available for experimentation.

The detailed literature review identified work carried out to resolve vibration problems associated with the fuel pumps of the space shuttle [13]. The adverse effects of fluid rotation on the stiffness and damping performance on the fine clearances in pumps were identified. Other investigators and analysts provide additional support data [14].

The initial plan involved reducing the annular seal clearances and applying plain surface profiles. As a backup, and in order to reduce the adverse effects of prerotation of the fluid prior to entry into the clearances, radial slots were to be introduced into the casing rings at the impeller eyes, and radial and axial slots were to be introduced into the balance drum bushing. The stiffness and damping values with different levels of prerotation are displayed in Table 2. For completeness, the values developed for the initial configuration are included. The adverse effect of the balance drum grooving is apparent from the negative value associated with the direct stiffness K_{xx}. The balance drum configuration developed for the zero swirl condition is shown in Figure 3.

Due to the complexity of incorporating the added mass effects into the stability program, values of the impeller/diffuser interaction forces calculated by the California Institute of Technology were not used. Instead, a generalized equation was utilized to simulate the impeller/diffuser forces to provide a relative value associated with the instability or stability of the various configurations expressed in terms of the logarithmic decrement [15]. To determine the sensitivity of the rotor to the destabilizing forces, the level of the destabilizing force was varied over a large range. However, variation of these forces was not required to identify instability. Some of the computed results are presented in Table 3. Logarithmic decrement data and associated instability frequencies indicate:

• Good agreement with the site data was achieved, indicating a negative logarithmic decrement (the definition of logarithmic decrement is shown in Figure 4 at zero impeller/diffuser interaction force, as a result of out of balance response).

• The stabilizing effects of the redesign are indicated by the high positive logarithmic decrement. It should be noted that for the ring configuration employed, an increase in clearance can be accommodated without severely affecting the value of the logarithmic decrement.

Table 2. Stiffness and Damping Factors.

ORIGINAL DESIGN

Impeller Neck Seals – Inlet Swirl = 0.7RΩ
K_{xx} = 5737 lb/in K_{xy} = 7600 lb/in C_{xx} = 16 lbsec/in
Interstage Seal – Inlet Swirl = 0.25RΩ
K_{xx} = 5570 lb/in K_{xy} = −1890 lb/in C_{xx} = 9.4 lbsec/in
Balance Drum – Inlet Swirl = 0.7RΩ
K_{xx} = −82400 lb/in K_{xy} = 153600 lb/in C_{xx} = 253 lbsec/in

MODIFIED DESIGN WITHOUT SWIRL BRAKES

Impeller Neck Seals – Inlet Swirl = 0.7RΩ
K_{xx} = 309400 lb/in K_{xy} = 16900 lb/in C_{xx} = 33.5 lbsec/in
Interstage Seal – Inlet Swirl = 0.25RΩ
K_{xx} = 10000 lb/in K_{xy} = 1920 lb/in C_{xx} = 13 lbsec/in
Balance Drum – Inlet Swirl = 0.25RΩ (Partial Swirl Brakes)
K_{xx} = 598000 lb/in K_{xy} = 1023000 lb/in C_{xx} = 1400 lbsec/in

MODIFIED DESIGN WITH SWIRL BRAKES

Impeller Neck Seals – Inlet Swirl = 0.25RΩ
K_{xx} = 19800 lb/in* K_{xy} = 1750 lb/in* C_{xx} = 12.2 lbsec/in
Interstage Seals – Inlet Swirl = 0.7RΩ
K_{xx} = 10000 lb/in K_{xy} = 1920 lb/in C_{xx} = 13.16 lbsec/in
Balance Drum – Inlet Swirl = 0 (Extended Axial Slots)
K_{xx} = 659000 lb/in K_{xy} = 586000 lb/in C_{xx} = 3000 lbsec/in

*Seal length reduced by 40% due to radial slots introduced as swirl brakes for configuration analyzed.

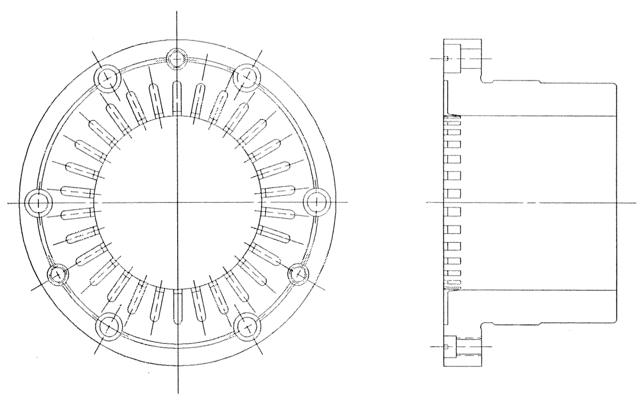

Figure 3. Balance Drum Bush—Zero Inlet Swirl Configuration.

Table 3. Rotor Stability.

	Original		Plain/Plain		Installed		2 × Installed	
	cpm	δ	*cpm*	δ	*cpm*	δ	*cpm*	δ
F	4056	−2.11	2820	3.89	2020	7.35		
F	5153	0.64	4867	0.53	4871	0.53	4858	0.52
B	5483	3.77	5301	4.07	5310	4.06	5393	4.05
F	13951	0.77	9533	0.40	9530	0.40	13949	0.69
B	13761	0.82	8840	1.18	8844	1.19	8862	1.28

Legend

F = forward whirl
B = backward whirl
δ = logarithmic decrement
Impeller/diffuser interaction force = 50000 lb/in

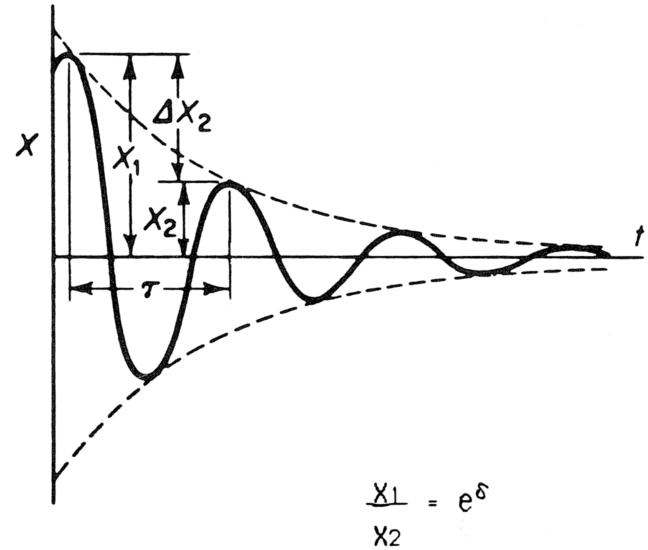

Figure 4. Definition of Logarithmic Decrement (δ).

TEST DATA

As previously discussed, physical modification to a machine was carried out in parallel with the analysis effort. The complete pumps had been shipped back to the manufacturing plant, based upon a decision made to institute a test program in the factory. This was considered advantageous, although the impact of the difference in pumped medium would remain an unknown until recommissioning on-site. In addition to using non-contacting displacement probes to monitor shaft orbits, pressure transducers were introduced into the suction bay, one diffuser, case drain, balance pipe and discharge main. This was done to detect or eliminate the rotating stall mechanism or acoustic resonances as possible causes of the rotor vibration. Axial displacement probes were applied to the non-drive end bearing housing, and the bearing housing position was monitored as the temperature of the pumped fluid was changed. Due to the apparent adverse effect of temperature during site operation, shop testing was conducted up to a temperature of 350°F. This was within the range of identified unstable operation.

It was considered that a reduction in clearance, and the elimination of the threading in the fine clearances, would be sufficient to stabilize the rotor. As a result, the machine was initially built for testing without the radial and axial slots. This configuration, however, proved to be unsuccessful. The subsynchronous vibration was observed at a high level, but with a frequency slightly below half of the running speed. The slot configuration was, therefore, considered necessary to effect a solution to the problem.

Following further modification, the unit test data exhibited no discrete vibration frequencies over the whole flow range. However, at flowrates close to the minimum flow of 300 gpm, high levels of noise at random frequencies were observed at the coupling end of the machine. An analysis of the machine design had already identified that the eye diameter and the angles of the vanes of the suction impeller had been over designed, and recirculation flow in the suction impeller was responsible for the random frequency disturbance [16]. Due to time constraints, it was not possible to redesign and manufacture a new suction impeller. To eliminate the random disturbance, a stage impeller was substituted for the suction impeller. This impeller had more conventional proportions, and there was sufficient net positive suction head (NPSH) available on-site to support the substitution. Subsequent testing indicated that the disturbance was eliminated, and the machine could operate at vibration levels of less than 1.0 mils peak-to-peak at 6600 cpm over the total flow range (Table 4).

Table 4. Pump Vibration Data.

SPEED RPM	DRIVE END X Unfil	DRIVE END X Fil	DRIVE END Y Unfil	DRIVE END Y Fil	FREE END X Unfil	FREE END X Fil	FREE END Y Unfil	FREE END Y Fil	FLOW GPM
Test Stand Vibration Levels									
6600	0.39	0.22	0.52	0.25	0.6	0.44	0.6	0.49	412
6600	0.32	0.21	0.44	0.18	0.61	0.45	0.64	0.50	430
6600	0.38	0.21	0.51	0.23	0.74	0.49	0.78	0.57	290
Site Vibration Levels									
6600	0.95	0.54	0.98	0.52	0.23	0.07	0.25	0.07	
6600	0.97	0.53	0.98	0.51	0.23	0.07	0.24	0.07	
6600	0.97	0.54	0.99	0.52	0.24	0.07	0.24	0.07	

All vibration levels in mils peak-to-peak measured by non-contacting displacement transducers on the pump shaft.

TEST DATA ON-SITE

Having completed shop testing, the final proof of an effective fix required successful recommissioning of the units on-site. The machines were comprehensively instrumented, and performances were monitored on an around-the-clock basis for the initial commissioning. Care was taken to ensure that correct warm-up procedures were applied to the units, and the pumps were gradually brought up to load and temperature. Site performance was similar to that experienced in the shop test program, and typical data supplied by the refinery inspectors may be seen in Table 4.

CONCLUSIONS

The data developed during original site commissioning, and the subsequent investigative test program, clearly demonstrates the impact of the fine internal clearances on the performance of the rotor in machines employing a large number of stages. The adverse effects of fluid rotation in these fine clearances is also clearly demonstrated.

Although not central to the problem, the effect of recirculation from the suction impeller on rotor stability was highlighted.

Analyses results indicate a marginal improvement on the rotor stability with worn internal clearances. This is contrary to normal expectations. However, in this case, the close clearance and relatively high speed promote significant fluid rotation as the flow passes through the seals. For the increased clearances, this potentially destabilizing effect is reduced. No test data is available to support this theoretical computation.

Agreement between the original site test data, associated with the failures and theoretical predictions, is encouraging. However, it is appreciated that much more development work is required in order to refine the science of calculating both the forced response and stability of multistage pump rotor systems.

NOMENCLATURE

- a — speed of sound in fluid (celerity)
- d — diameter of passage
- f — organ pipe frequency
- g — acceleration due to gravity
- t — thickness of passage wall
- K — bulk modulus of fluid
- B — backward whirl direction
- E — Young's modulus for passage wall material
- F — forward whirl direction
- L — length of flow passage
- N — running speed
- R — radius from center line of shaft
- X — vibration displacement
- δ — logarithmic decrement
- γ — specific weight of fluid
- Ω — rotational speed
- X_1 — vibration displacement
- X_2 — vibration displacement
- C_{xx} — direct damping coefficient of annular seal
- K_{xx} — direct stiffness coefficient of annular seal
- K_{xy} — cross-coupled stiffness coefficient of annual seal

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